1989

Pressure drop behavior of saturated water /

Sugeng Sundjaswadi
Lehigh University

PRESSURE DROP BEHAVIOR OF

SATURATED WATER

by

SUGENG SUHDJASWADI

A THESIS

Presented to the Graduate Faculty
of Lehigh University
in Candidacy for the Degree of
Master of Science

Lehigh University

Bethlehem, Pennsylvania

1955

CERTIFICATE OF APPROVAL

This thesis is accepted and approved in partial fulfillment of the requirements for the degree of Master of Science in Chemical Engineering.

Mar 22, 1955
Date

Alan S. Foust
Professor in Charge

Alan S. Foust
Head of the Department

ACKNOWLEDGMENTS

This problem was proposed and carried out at Lehigh University under the supervision of Dr. A. S. Foust to whom the author expresses his appreciation for his helpful criticism and frequent suggestions.

The author also wishes to express his gratitude to the Institute of Research of Lehigh University for undertaking the cost of equipment.

Finally, the author wishes to thank the staff members of the Chemical Engineering Department of Lehigh University for the help they have provided as teachers and to acknowledge the assistance of W. Szulborski with the construction of the equipment.

Sugeng Sundjaswadi

Sugeng Sundjaswadi

TABLE OF CONTENTS

ABSTRACT

This report presents the results of an investigation on the pressure drop behavior of saturated water. Copper tube of 0.495 inside diameter (O.D.-0.625 in) was used. The test tube was intentionally small for measuring point conditions.

The mass flow rate ranged from 500 - 1300 lb/ft^2-sec. and the liquid was carefully checked to be at the saturation point in the mixing tank.

The results show that the largest pressure drop is encountered near the end of the pipe. Further it is noticed that in the range of mass flow rate used, the pressure drop over the entire length does increase with increase of mass flow rate.

An effort was made to explain the phenomenon of sharp pressure drop near the end of the pipe using combination of thermodynamic theory and theory of dimensions. Equations are given which shows that vapor formation near the end of the pipe is tending to be isentropic.

INTRODUCTION

In many industrial heating processes, hot water at a temperature corresponding to its boiling point (termed conveniently : saturated water) is handled through valves and pipes, traps and pipes. Examples are blow down from a boiler, the flow of saturated water to and from steam traps, the flow of condensate through piping and traps, when steam bled from the various stages of steam turbines is used for heating feed water. In such cases, reasonable knowledge of laws of flow of hot water near or at its saturation point is necessary.

Furthermore, much attention and effort have been given in recent years toward obtaining better understanding of the properties of boiling liquids. The motivation of this emphasis can be sought in the trend of cooling of high performance engine by utilizing satured liquid with its high heat transfer coefficient and constant surface temperature. One of the important properties which has a bearing on the heat characteristic sought is the pressure and the pressure drop behavior of the saturated liquid. It is a known fact that the heat transfer coefficient increases with increase of velocity, but the accompanying increase of pressure drop may off-balance the gain in the heat transfer characteristic.

The knowledge of pressure drop behavior of saturated liquids is also of importance for the stability of water distribution in heating surfaces of forced flow boilers. Previous investigations led to the conclusion that deviation from uniform flow distribution will be minimized if the pressure drop increases with the flow rate G.

It is clear that pressure drop behavior of saturated liquid is one of its important properties and knowledge of it is essential.

The purpose of this investigation is to determine and explain the flow characteristics of saturated water through pipes and devise a method for predicting its mass rate.

LITERATURE SURVEY

Previous investigations concern most about flow of saturated water through nozzles, orifices and short tubes. In general experimental data present indicate the supposedly maximum flow rate for various initial pressures and do not give an indication of the flow characteristic. An exception is R. S. Silver (12,13).

One of the first investigators was Prof. Rateau (14), who showed that for any given initial saturation pressure, there is a critical throat pressure for which the discharge is a maximum and that no increase of flow occurs for any decrease of back pressure below this critical pressure, just as for the flow of steam. Bottomley (3) obtained measured maximum discharges which are from four to five times as great as would be expected from ordinary thermodynamic theory. Bottomley based his calculation of maximum discharges on the thermodynamic equilibrium in the orifice. Assuming constant entropy (isentropic) expansion he arrived at values which are one fifth to one fourth of the actual discharge. Benjamin and Miller (1) gave as conclusions of their investigations on flow of saturated water through orifices, that the discharge coefficients found are approximately the same as those generally used for cold water. Data obtained in their investigations does not reveal the presence of critical pressure in the orifice. Burnell (4) gave as reason for the fact that discharges through nozzles are four to five times those calculated from the thermodynamic theory the delay in vaporization of water, allowing a greater pressure drop without evaporation until the throat is reached. Equations for flow of boiling water through a nozzle derived from entropy-temperature diagram is given in his paper.

In addition to his investigations of flow of saturated water through nozzle, Bottomley (3) formulated expressions covering the flow in pipes following a nozzle. For all his calculations he used a coefficient of friction K of 0.012. Benjamin and Miller (2) carried out similar measurements of the flow of saturated water and came to the same conclusions that saturated water discharge through a pipe will depend, in general, on the initial pressure and not on the receiver back pressure. Their value for coefficient of friction K is between 0.0116 - 0.0131, almost in agreement with Bottomley's result. Burnell (4) however used as friction factor coefficient a value of 0.018, this being the approximate mean value of K for the Reynolds numbers corresponding to the range of velocities and temperatures of both steam and water in his experiments. His measured and calculated values for discharge and exit pressures are not in agreement. By assuming that the saturated water and its steam velocities are not identical (assumption:

velocity of water = $\frac{1}{1.75}$ of its steam), his computed values were then almost in accordance to measured values. Schweppe and Foust (10) have conducted an additional investigation on the behavior of pressure drop of boiling water and found that it passes through a maximum value for velocity of 3 ft/sec beyond which it first decreases and afterwards increases again with increasing mass rate of flow. They came to the conclusion that the maximum value for pressure gradient is due to sonic velocity at the end of the pipe; Isbin (6) was doubtful about this conclusion, however without giving his analysis. Ledinegg (8) and Profos (9) suggested that pressure drop decrease with increase of flow rate for a certain flow rate interval will be found where heat absorption causes generation of steam, creating volume increase of the flow medium. As safe practical speed Ledinegg suggested velocity beyond 3 ft/sec, which should be high enough to carry away any steam bubbles.

THEORY

When saturated liquid flows in a pipe of uniform cross section the mass flow rate G (lb/sq.ft.sec.) is constant throughout the pipe length. The velocity at any point increases and is a function of the specific volume. Computations can be carried out without undue difficulties only under definite assumptions.

The most important assumption made in this investigation was that the energy of the liquid remains constant during the flow process i.e. that no heat be added or conducted away. Therefore the system (especially the test and the calming section) were well insulated.

In deriving the following equations, following assumptions has been made:

1. The properties of flowing liquid are uniform at any cross section.
2. The properties vary from point to point according to equilibrium.

It is thus assumed that liquid and the vapor "in solution" move with the same velocity.

If a saturated liquid flows adiabatically in a pipe, pressure drop causes also a decrease of the saturation pressure accordingly, the energy liberated by this adiabatic expansion will be expended partly in overcoming the friction and partly in increasing the kinetic energy.

Around a flow system, energy balance between two adjacent cross section of a pipe, where no accumulation of energy occurs, is given in general as follows:

$$E_1 + \frac{mU_1}{2g_c} + \frac{mgZ_1}{g_c} + P_1V_1 + Q - W = E_2 + \frac{mU_2^2}{2g_c} + \frac{mgZ_2}{g_c} + P_2V_2$$

By definition: $\Delta E = E_2 - E_1$

$$\Delta PV = P_2V_2 - P_1V_1$$

$$\text{Therefore } \Delta E + \Delta \frac{mU^2}{2g_c} + \Delta \frac{mgZ}{g_c} + \Delta PV = Q - W \qquad (1)$$

The increase in E is the sum of the increases due to all changes as taking place in the material in flow, including heat effects, compression (expansion) effects, surface effects and chemical effects:

$$\Delta E = \int_1^2 TdS + \int_1^2 PdV + \int_1^2 \gamma\, d\sigma + \int_1^2 \underset{A}{u}dm_A + \text{etc} \qquad (2)$$

The increase due to heat effects Td S is equal to the sum of the heat absorbed from the surroundings and all other energy dissipated into heat effects within the system due to irre-versibilities in the process:

$$Td\ S = Q + (lw) \qquad (3)$$

where lw = "lost work", energy that could have done work but was dissipated in irreversibilities within the flowing material.

Substituting (2) and (3) in (1), and neglecting surface and chemical effects the following equation will be obtained:

$$VdP + \frac{mU^2}{2g_c} + \frac{mgZ}{g_c} = -W \ -lw \qquad (4)$$

Equation (4) is not dependent upon whether heat is transferred, but it has the term (lw), representing irreversibilities, all of which has to be included herein.

In order to evaluate incomplete differentials, such as VdP, appearing on the other side of the equation, the exact path followed by the flowing material has to be known.

In the particular case of saturated water flowing in a pipe, where adiabatic process is assumed, in which no work (external) is done and no change in elevation takes place, equation (4) may be written as:

$$VdP + \left(\frac{m\,U^2}{2\,g_c}\right) = -lw \qquad (5)$$

As indicated lw includes friction losses and other irreversibilities; in the case of adiabatic flow it will require only a knowledge of the frictional losses, which can be determined by actual experiment.

Replacing lw by dF and rewriting the flow process for a unit mass of material, we will have the following equation:

$$VdP + \left(\frac{U^2}{2g_c}\right) + F = 0 \qquad (6)$$

$$\text{or:} \int_1^2 VdP + \int_1^2 \frac{UdU}{g_c} + \int_1^2 dF = 0$$

$$\text{As} \quad dF = \frac{U^2\,K\,dL}{2g_c\,D} \qquad (7)$$

Equation becomes:

$$\int_1^2 VdP + \int_1^2 \frac{UdU}{g_c} + \int_1^2 \frac{U^2 KdL}{2g_c D} = 0 \qquad (8)$$

K = "friction factor" corresponding to Reynolds numbers. In steady state flow problems in a conduit, the basis for thermodynamic analysis is given by the continuity equation, which gives the relationship between mass rate of flow, w, the velocity, specific volume and the cross sectional area of the conduit A:

$$w = \frac{U_1 A_1}{V_1} = \frac{U_2 A_2}{V_2} \qquad (9)$$

The subscripts refer to any section of the conduit. For a pipe of constant cross section area this equation is written as:

$$\frac{w}{A} = \frac{U_1}{V_1} = \frac{U_2}{V_2} = G \text{ (Mass Flow Rate)} \qquad (10)$$

or: $V\,G = U$ (11)

$GdV = dU$ (12)

Therefore $UdU = VG^2 dV$ (13)

Substituted in (8) will give:

$$\int_1^2 VdP + \int_1^2 \frac{G^2}{g_c} VdV + \int_1^2 \frac{KG^2\, V^2 dL}{2g_c D} = 0 \qquad (14)$$

or:

$$\int_1^2 \frac{dP}{V} + \int_1^2 \frac{G^2 dV}{g_c V} + \int_1^2 \frac{KG^2\, dL}{D\, 2g_c} = 0 \qquad (15)$$

Replacing V by $1/\rho$ and rearranging:

$$\int_1^2 \rho \, dP - \frac{G^2}{g_c}\int_1^2 \frac{d\rho}{\rho} + \frac{G^2 K}{2g_c D}\int_{L_1}^{L_2} dL = 0 \qquad (16)$$

The second and the third term are directly integrable.

If P = lb/sq.in, then:

$$G^2 \ \ln \left(\frac{\rho_1}{\rho_2}\right) + \frac{G^2 KL}{D} = 144 \times 2 \ g_c \int_{P_1}^{P_2} \rho \, dP \qquad (17)$$

The values for the densities in the above equation depends upon the proportion of vapor formed at each cross section during the flow. As the specific volume of vapor is so large, compared with the specific volume of liquid, even a very small proportion of vapor formed will cause an appreciable change in the specific volume (density) of the fluid as a whole.

It is felt that determination of the amount of vapor formed cannot be analyzed and predicted with the usual thermodynamic theory as many factors interconnected with each other, have a bearing on vapor formation.

However, some useful indications may be deduced by using the theory of dimensions. Dimensional treatment will be here particularly valuable, as the mathematical relations are unknown.

Considering q as the quantity to be investigated, variables which have effect on this q are then as follows:

V_f = Volume of vapor per pound.

V_g = Volume of liquidper pound.

λ = Latent heat of evaporation per pound.

G = Mass flow rate.

L = Length of the tube section concerned.

D = Diameter of the tube.

u = Dynamic viscosity of fluid in flow.

We may represent q as a function of these variables of the form:

$$q = \phi\,(V_f^a,\ V_g^b,\ \lambda^c,\ L^d,\ D^e,\ G^f,\ u^g) \tag{18}$$

The dimensions of the various variable quantities are as follows: (M, L, θ units):

$V_f = L^3/M$

$V_g = L^3/M$

$\lambda = L^2/\theta^2$

$G = M/\theta L^2$

$L = L$

$D = L$

$u = M/\theta L$

Hence the dimensions of the functions ϕ are:

$$L^{3a}\,M^{-a}\,L^{3b}\,M^{-b}\,L^{2c}\,\theta^{-2c}M^{f}\,\theta^{-f}L^{-2f}\,L^{d}\,L^{e}\,M^{g}\,\theta^{-g}\,L^{-g} =$$

$$L^{3a+3b+2c+d+e-2f-g}\quad M^{-a-b+f+g}\quad \theta^{-2c-f-g} \tag{19}$$

But as q is dimensionless number (lb/lb = M/M = 1)

$$0 = 3a+3b+2c+d+e-2f-g \tag{20}$$

$$0 = -a-b+f+g \tag{21}$$

$$0 = -2c-f-g \tag{22}$$

There are altogether three equations and seven variables or unknowns, as such only three may be solved. Treating a, b, d and f as knowns, we can compute c, e and g in terms of the known exponents, as follows:

from (21) $\quad -f-g = -a-b$

from (22) $\quad +2c = -f-g = -a-b \quad$ or $c = \frac{-(a+b)}{2}$ (23)

$$-g = -(a+b)+f$$

$$\text{or } g = +(a+b)-f \tag{24}$$

Substituted in (20) we will have:

$$3a+3b-a-b+d+e-2f-a-b+f = 0$$

$$\text{or } a+b+d+e-f = 0$$

$$e = -(a+b)-d+f$$

Therefore:

$$q = \phi \; V_f^a \, V_g^b \, \lambda^{\frac{-(a+b)}{2}} \, L^d \, D^{-(a+b)-d-f} \, G^f \, u^{+(a+b)-f} \tag{25}$$

Dimensionally we cannot distinguish between V_f and V_g; we therefore can conclude that we have some linear function of V_f and V_g to the power $(a+b)$; in our case the difference $V_f - V_g$ is the linear function which has physical meaning.

The following correlation can be obtained:

$$q = \phi \; (V_f - V_g)^{a+b} \, \lambda^{\frac{-(a+b)}{2}} \, L^d \, D^{-(a+b)-d-f} \, G^f \, u^{(a+b)-f} \tag{26}$$

Rearranging it, the following groups will appear:

$$q = \phi\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{a+b} \left(\frac{L}{D}\right)^{d} \left(\frac{G\,D}{u}\right)^{f} \qquad (27)$$

Thus q is a function of three non dimensional functions namely:

$\left(\frac{L}{D}\right)$ = which can be called the shape number

$\left(\frac{G\,D}{u}\right)$ = which is the Reynolds number

$\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right) = N_p$ = which depends upon the condition of the fluid in flow (saturation pressure and temperature)

According to this dimensionless groupings, the q formed/lb of saturated liquid is a function of the ratio L/D, Re-number, and a group representing the physical properties of the fluid in flow, depending upon the saturation pressure and temperature.

This equation and thermodynamic theory combined will give solution of the actual q formed.

For a definite pressure drop, isentropic expansion can be easily computed, as the proportion of the vapor formed is then only dependent upon the physical properties of the liquid at that saturation point and is independent of the velocity (Re-number), pipe dimensions (L/D) and consequently the friction factor. The densities can then be calculated directly and by means of equation (17), whereby the friction factor is neglected, the mass flow rate can be obtained at any cross section. But naturally enough the calculated values are not in accordance with the experimental values as the pipe and fluid friction amongst other factors for instance, will impose an action which causes the expansion (amount of vapor formed) to depart to some extent from the isentropic.

As q_{isen} is only dependent upon N_p we can write:

$$q_{isen} = \phi \left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{(a+b)} \tag{28}$$

In actual case, some constraint is applied to the fluid in flow; the actual q formed is then a function as given in the equation (27) but the factor N_p will be the same as in the case of isentropic expansion, as the point at which the fluid is computed is identical (same saturation pressure and temperature) or:

$$q_{act} = \phi \; (L/D)^d \; (R_e)^f \; \left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{a+b} \tag{29}$$

Therefore we can conclude that:

$$\frac{q_{isen}}{q_{act}} = \phi \; \frac{1}{(L/D)^d \; (R_e)^f} = \phi \; (L/D)^{-d} \; (R_e)^{-f} \tag{30}$$

In equations (16) and (17) it will be noticed that the mass flow rate G (or rather G^2) is in some way related to the density, which in turn is dependent upon the q formed.

As such it can be said that G^2 is a function of q or:

$$G^2 = \phi\,(q) = \phi\,\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{z}$$

$$\text{or} \qquad (q) = \phi\,\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{a+b+z} G^{-2} \tag{31}$$

For isentropic expansion this will give then:

$$q_{isen} = \phi\,\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{a+b+z} G_{isen}^{-2} =$$

$$\phi\,\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{r} G_{isen}^{-2} \tag{32}$$

For actual flow:

$$q_{act} = \phi\,(L/D)^{m}\,(R_e)^{n}\,\left(\frac{(V_f - V_g)u}{\lambda^{1/2} D}\right)^{r} G^{-2} \tag{33}$$

Again:

$$\frac{q_{isen}}{q_{act}} = \phi\,(L/D)^{-m}\,(R_e)^{-n}\,\left(\frac{G^2}{G_{isen}^2}\right) \tag{34}$$

DESCRIPTION OF APPARATUS

1. Test setup. (Fig. I)

The setup consisted of a flash tank (c), in which saturated water after passing the test section would be flashed at approximately 1 atmosphere. A centrifugal pump for recirculating the liquid was used (e), where after by means of a modified T-ejector (f) steam was introduced from the main line. The hot liquid passed the flowrator into the mixing tank (g). Condition of the liquid in the mixing tank would be indicated by the installed mercury (h) thermometer and the pressure; the saturated liquid was then forced through the calming section (b), which was a pipe of similar metal and dimension, prior to the test tube (a).

TABLE I

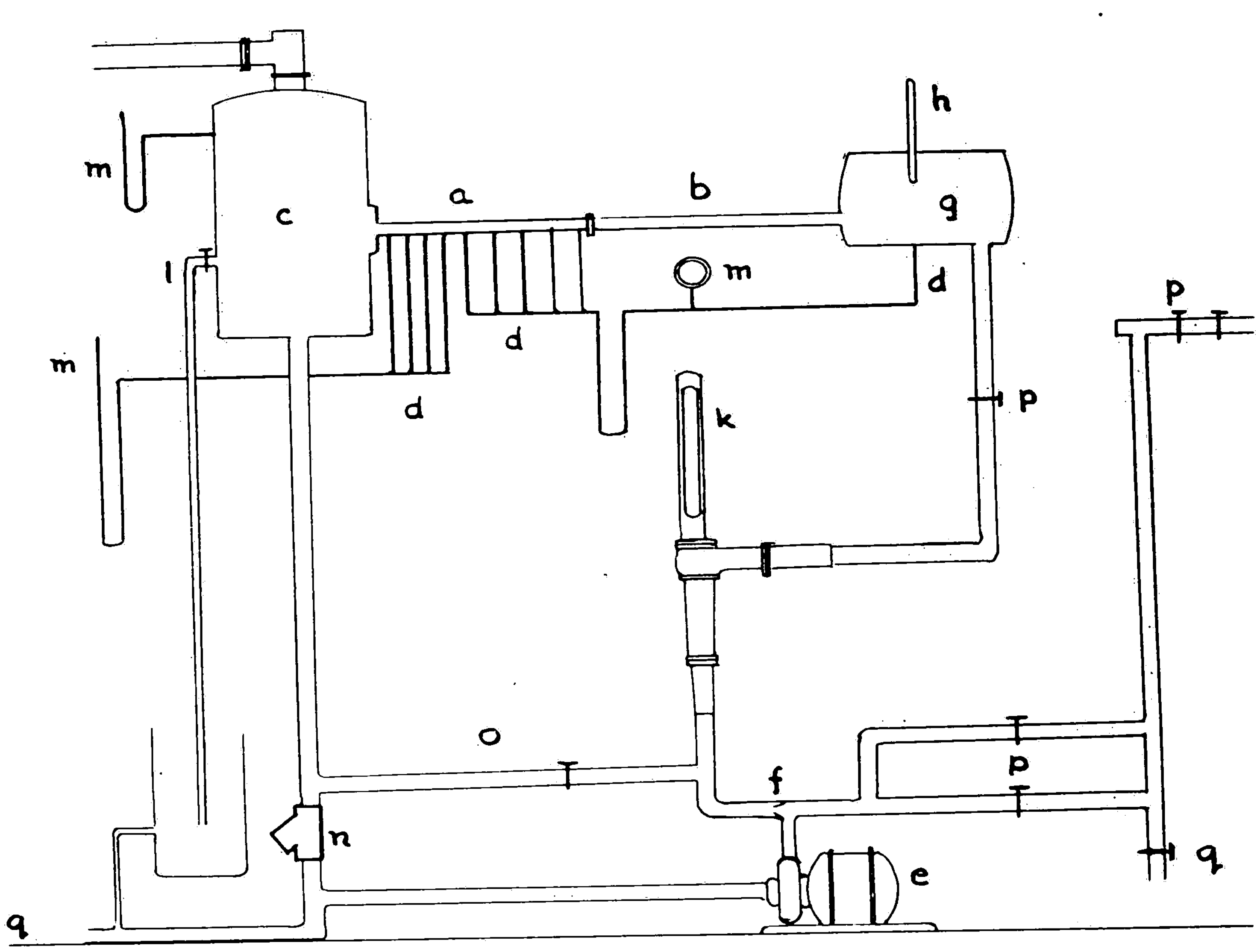
Dimensions of Test Tube

(All lengths in inches)

Material	Copper
Total length	36"
Calming section length	21"
Test section length	15"
Actual I.D.	0.495"
Actual O.D.	0.675"
S.P.S.	3/8"

FIG.1 DIAGRAM. OF EQUIPMENT.

a. TEST TUBE
b. CALMING SECTION
c. FLASH TANK
d. PRESSURE TAPS
e. CENTRIFUGAL PUMP
f. T - EJECTOR
g. MIXING TANK
h. THERMOMETER
k. FLOW RATOR
l. OVERFLOW
m. MANOMETER
n. STRAINER
o. BYPASS
p. CONTROL VALVES
q. TO DRAIN



2. Pressure-measurements. (Fig. I)

To measure the pressures, simple U manometers of glass and a Bourdon gage were employed (m). Nine very small holes were drilled for the test tube at intervals of 2-1/2" for the first five taps and 1" for the other 4 taps. The first pressure taps was about 9/16" from the inlet and the last one was 7/16" from the outlet end, which was due to the fact that the test tube was threaded at both ends (1/22 and 3/8" respectively). 1/16 " tubing was welded on to connect to the manometer. To secure minimal disturbances of flow rough ridges were removed. Pressure of the liquid in the mixing tank was measured with a calibrated Bourdon gage which with the thermometer, placed in the tank, indicated the condition of saturation of the flowing water.

The first five pressure taps of the section is communicated by a manifold through a tube connection with one leg of one of the U tube mercury manometers, the other communicating with the pressure of the mixing tank, giving then the pressure drop towards the saturation pressure of the tank.

The last four pressures were measured by the other U-tube mercury manometer, whose other leg is communicated with the atmosphere, so that the difference between atmospheric and the pressure to be measured was indicated. The absolute pressure was the sum of the difference read and the barometer pressure.

3. Flow rate measurements. (Fig. III)

Boiling liquid to the test section was accomplished by heating the water with injected steam. Earlier test was done by using the laboratory boiler (Mears/Kane Ofeldt, 5 H.P. 50 psia-gas fired boiler), but it appeared that the steam produced was fluctuating considerably. Steam from the high pressure main was then used, which furnished enough constant steam supply to achieve saturation point in the mixing tank.

A self-designed T-injector was used in the line, as the Mac Dougall Tee-injector previously installed gave slugging effect due to the fact that the outlet-opening of the ejector was too large.

The liquid circulation was measured with a Fisher & Porter metal tube flowrator (1-1/2" inlet and outlet opening). Type used was 52-10, having flanged connections with the whole piping system. It was provided with a magnetic indicating extension, comprising of a nonmagnetic extension well, armature which was attached to the float in the flowrator by an extension rod, a follower closely fitting the outside diameter of the extension well.

The follower, attracted by the armature with the change in float position, would slide vertically and its position as read on the scale located on the glass tube indicated the flow rate through the flowrator.

The flowrator was calibrated before installing in the system and was found in agreement with the calibrated values from the company.

4. Test tube.

The test section proper (Fig. II - A) was a simple, smooth brass cylinder pipe, 15" in length, and adjoined on the inlet side by a similar pipe by means of a union (B). This was a part of a liquid recirculating and reheating system. This adjoining pipe was meant as a calming section and was 21" long. At the other end it was screwed to a threaded cap (2-1/2" diameter) (C) which in turn was screwed to a coupling (D) (1" long, 2-1/2" diameter) welded to the flash tank (E). At definite intervals along the pipe static pressure-holes (F) were located at which pressure is transmitted through the tubes to the manometer.

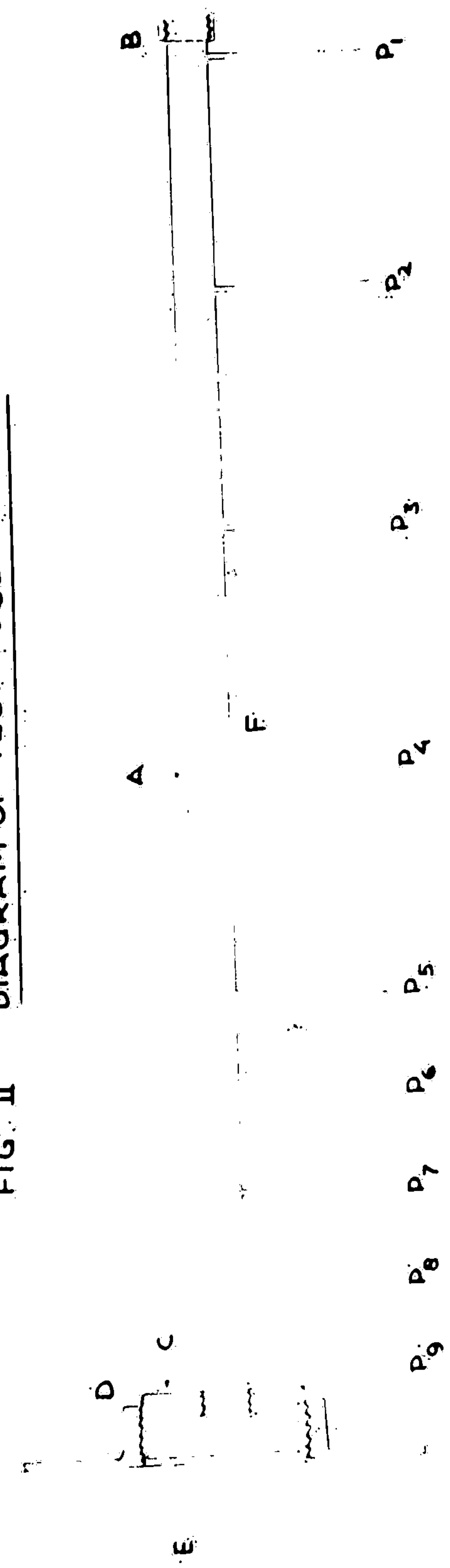
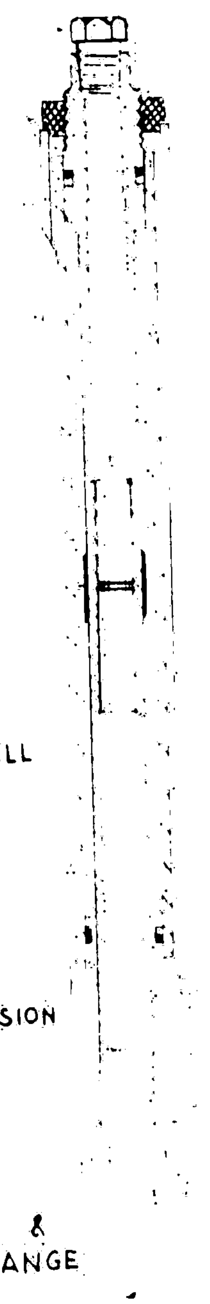


FIG. II DIAGRAM OF TEST TUBE SECTION

FIG. III DIAGRAM OF MAGNETIC INDICATING EXTENSION



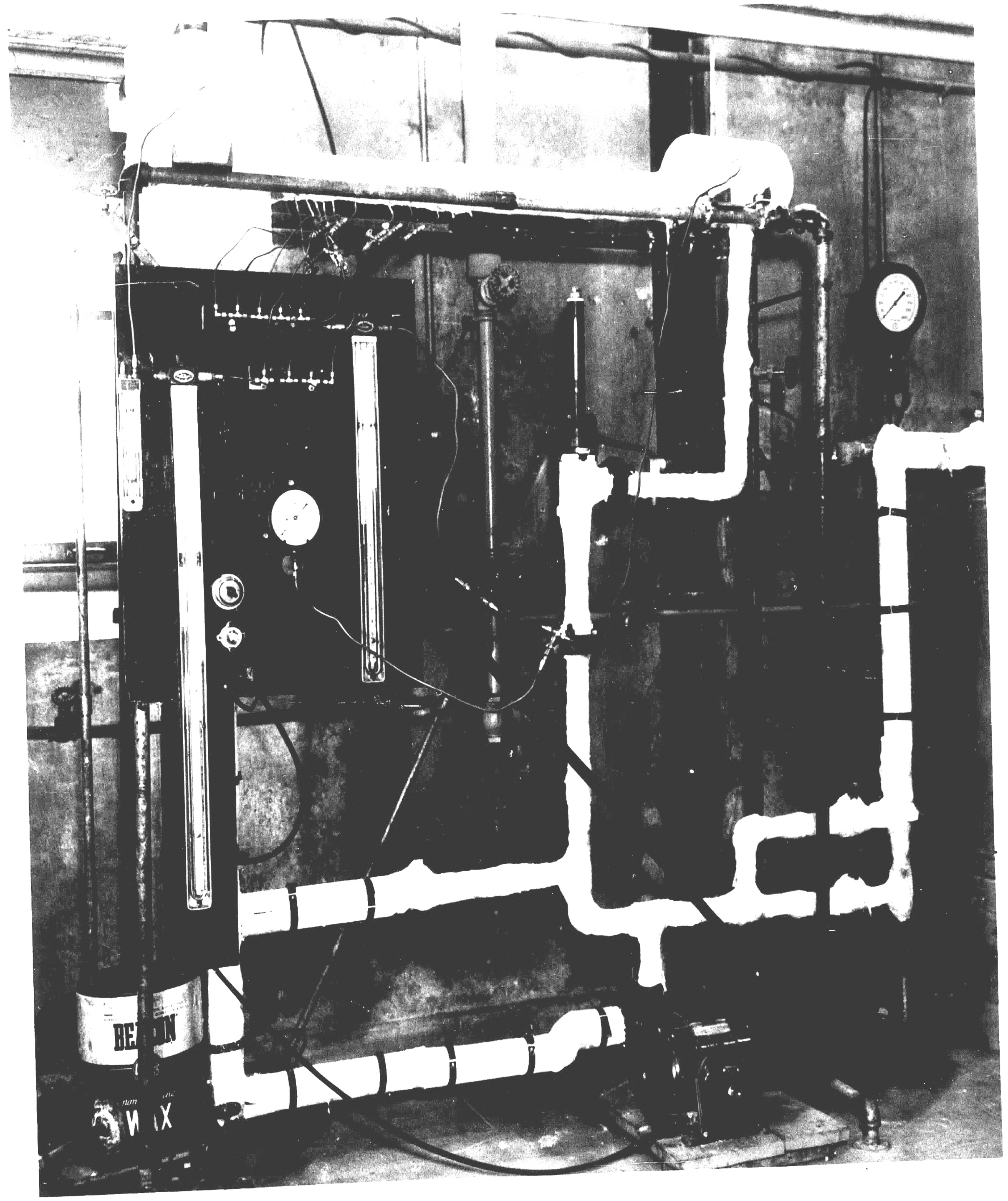


FIG. IV - APPARATUS

DISCUSSION OF THE RESULTS

The calculated results of this investigation are presented in Table II to V ; they are also presented graphically in Fig. V to VII.

Sample of calculations are given in Appendices B to G and Table VI to VII.

Table II represents the absolute pressure at different saturation conditions. Results of some of the runs are plotted in Fig. V where it could be noticed that the largest pressure drop occurs near the end of the pipe. Appendix B is a sample of calculation to arrive at the absolute pressure values for different pressure taps for each run.

Pressure drop data for the entire length were examined for different rate of flow and Fig.VI indicates that the pressure drop for the entire length increases as the mass rate is raised.

Results of ten (10) runs were taken at random to compute the amount of vapor formation if isentropic expansion is assumed and at the same time the actual vapor formed were calculated at conditions given by the experimental values of pressures. Table III gives the result of these computations; sample of method to arrive at the values is given in Appendix C with the remark that for actual vapor formation, values are assumed by trial and error, such that using KL/D at the condition concerned, equation (17) will give identical G value as the measured rate flow.

K values for the Reynolds number, corresponding to the range of velocity and temperature (pressure) of saturated water are given in Appendix D, whereby for viscosity data from Sigwart (11) and Heydweiller (5) are used. (Appendix E - Fig. VIII).

In the case of computation of isentropic vapor formation, values of entropy for saturated liquid and evaporation, specific volumes for saturated liquid and vapor from Keenan & Keyes are plotted against the pressure in order to facilitate extrapolation. This is given in Appendix F - Figs. IX, X.

From the result computation given in Table III it could be seen that the G_{isen} is increasing with increasing distance from the entrance or that the mass rate flow is tending to be isentropic near the end of the pipe.

To evaluate the exponents in the equation giving the relationship between $\frac{q_{isen}}{q_{act}}$ with the different non dimensional groups ($\frac{L}{D}$, R_e, $\frac{G}{G_{isen}}$) values of these groups are given in Table IV for 24 points of the 10 runs mentioned above. Table V present the Y-values, assuming the exponent for $\frac{L}{D}$ respectively 0.1, 0.333, 0.5, 1.0. These values are plotted in Fig. VII a, b, c, d. It will be noticed that

Fig. VII c gives the best correlation, the exponent for $\frac{L}{D}$ being 0.5. To arrive at the best line through these points method of curve-fitting is used, computation of which is given in Appendix G.

Equation (34) becomes then:

$$\frac{q_{isen}}{q_{act}} = 0.00375 \times \left(\frac{L}{D}\right)^{0.5} \times (R_e)^{0.4} \times \left(\frac{G}{G_{isen}}\right)^{2}$$

Remembering that $\left(\frac{G}{G_{isen}}\right)$ is decreasing with increase of distance from the entrance, the decrease of which is the largest near the end of the pipe, the conclusion can be taken that near the end of the pipe there is a tendency to isentropic expansion.

The phenomenon of sharp pressure drop near the end of the pipe can then be explained by the fact that the expansion of the saturated water is tending to be isentropic.

TABLE II ABSOLUTE PRESSURES AT DIFFERENT PRESSURE TAPS

FOR DIFFERENT MASS RATE OF FLOW

RUN	G	P_f	P_0	P_1	P_2	P_3	P_4	P_5	P_6	P_7	P_8	P_9
IV	1250	31.7	67.5	56.12	55.82	-	55.12	54.77	54.62	54.02	52.74	49.0
V	1395	31.7	67.5	54.92	54.62	54.52	54.12	53.92	53.49	53.19	52.24	49.94
VI	1062	31.7	59.4	52.8	52.0	-	50.4	50.0	48.22	47.74	44.64	43.84
VII	902	31.2	56.7	51.68	51.45	51.35	50.6	50.2	49.2	48.54	47.44	45.44
VIII	1062	29.78	63.38	54.53	54.28	54.18	53.33	52.78	-	52.47	51.17	49.52
9	809	30.78	41.2	38.1	38.04	38.0	37.58	37.3	36.62	35.82	34.98	33.45
11	589	30.6	40.7	40.16	39.86	39.52	38.58	38.18	37.18	36.68	35.72	34.22
12	652	31.0	44.22	43.0	42.7	42.47	41.77	41.04	40.94	40.34	39.4	37.67
13	872	31.6	53.4	48.57	48.39	48.32	47.95	47.39	46.39	45.69	44.29	42.59
15	634	31.14	41.34	40.79	40.54	40.22	39.22	38.97	38.08	37.42	36.69	35.23
16	872	31.84	52.80	48.51	48.31	48.19	47.71	47.23	46.05	45.55	44.30	42.30
17	954	31.6	60.72	53.65	53.45	53.45	53.1	52.4	51.94	51.34	50.44	47.64
18	965	31.2	57.1	48.85	48.75	48.65	47.95	47.65	46.29	46.09	45.04	42.59
19	1110	31.3	59.4	49.6	49.1	49.05	48.5	48.2	47.69	47.34	46.27	43.91
20	799	30.7	53.3	50.02	49.8	49.7	48.9	48.16	47.24	46.64	45.18	43.12
21	770	31.2	41.1	38.53	38.43	38.49	38.05	37.29	35.66	35.22	34.38	33.0
22	668.3	30.85	44.0	42.91	42.586	42.34	41.69	40.91	39.44	38.79	38.04	36.33
23	770	31.15	47.5	44.81	44.62	44.44	43.94	43.25	41.92	41.32	40.32	38.44
24	947.5	31.55	55.45	49.15	49.10	48.87	48.41	48.07	47.49	46.95	45.84	43.49
25	1271	31.07	65.2	52.51	52.24	52.51	51.74	51.41	51.21	50.81	50.21	47.81
26	1280	31.6	67.2	54.64	54.19	54.11	54.16	53.06	52.94	52.49	51.36	48.64
27	1045	31.7	55.7	48.4	48.18	48.0	47.62	47.32	46.38	45.84	44.52	42.04
28	978	32.0	53.9	46.17	48.12	47.97	47.54	47.12	46.24	45.64	44.54	42.28
29	1118	32.0	62.6	53.24	52.94	52.85	52.24	51.91	51.79	51.24	49.94	47.94
30	596	30.98	41.03	37.47	37.26	36.98	36.50	35.93	34.93	34.52	33.75	32.68
31	884	31.68	49.4	46.44	45.34	45.17	44.71	44.34	43.22	42.62	41.45	39.42

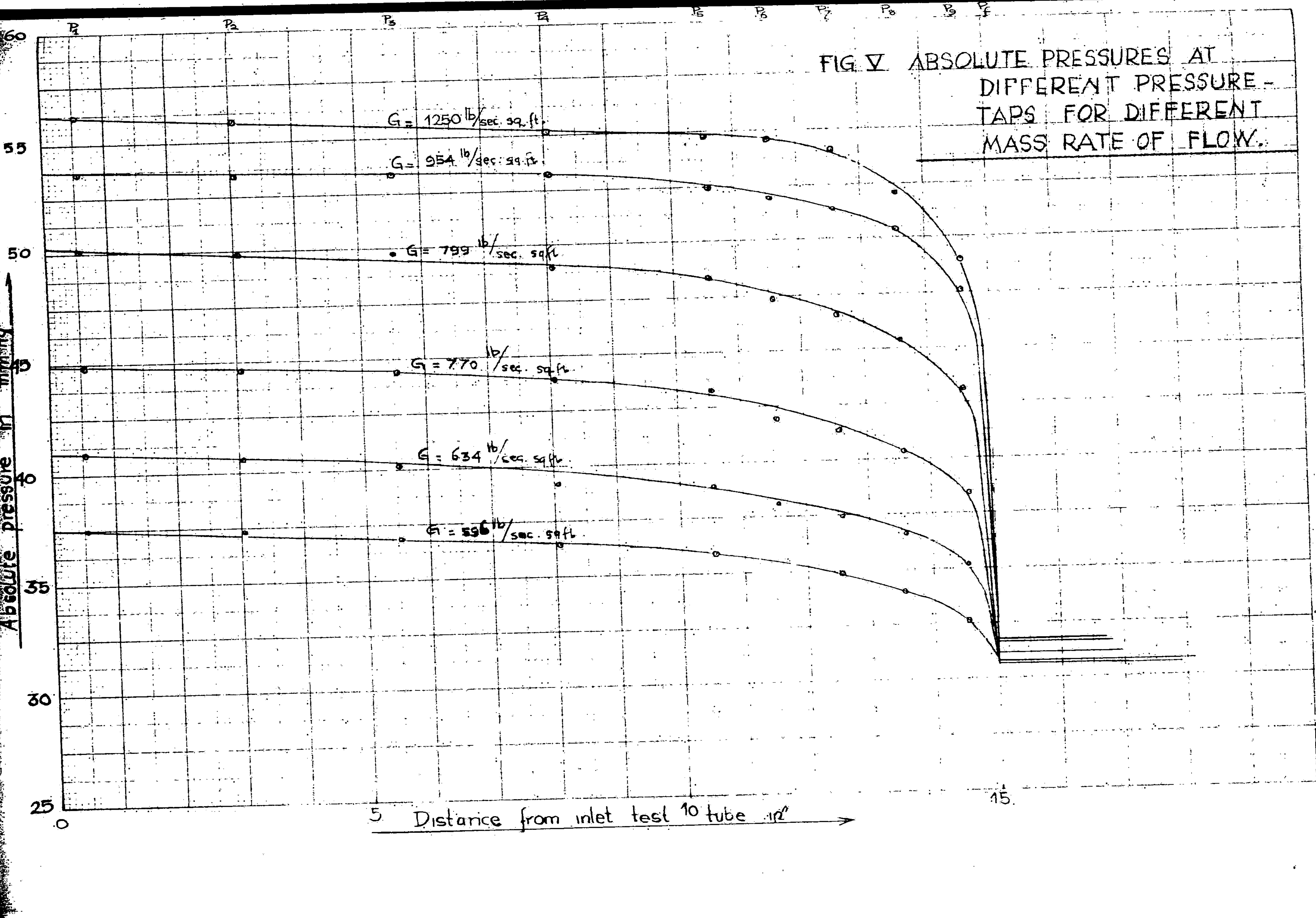


FIG V. ABSOLUTE PRESSURES AT DIFFERENT PRESSURE-TAPS FOR DIFFERENT MASS RATE OF FLOW.

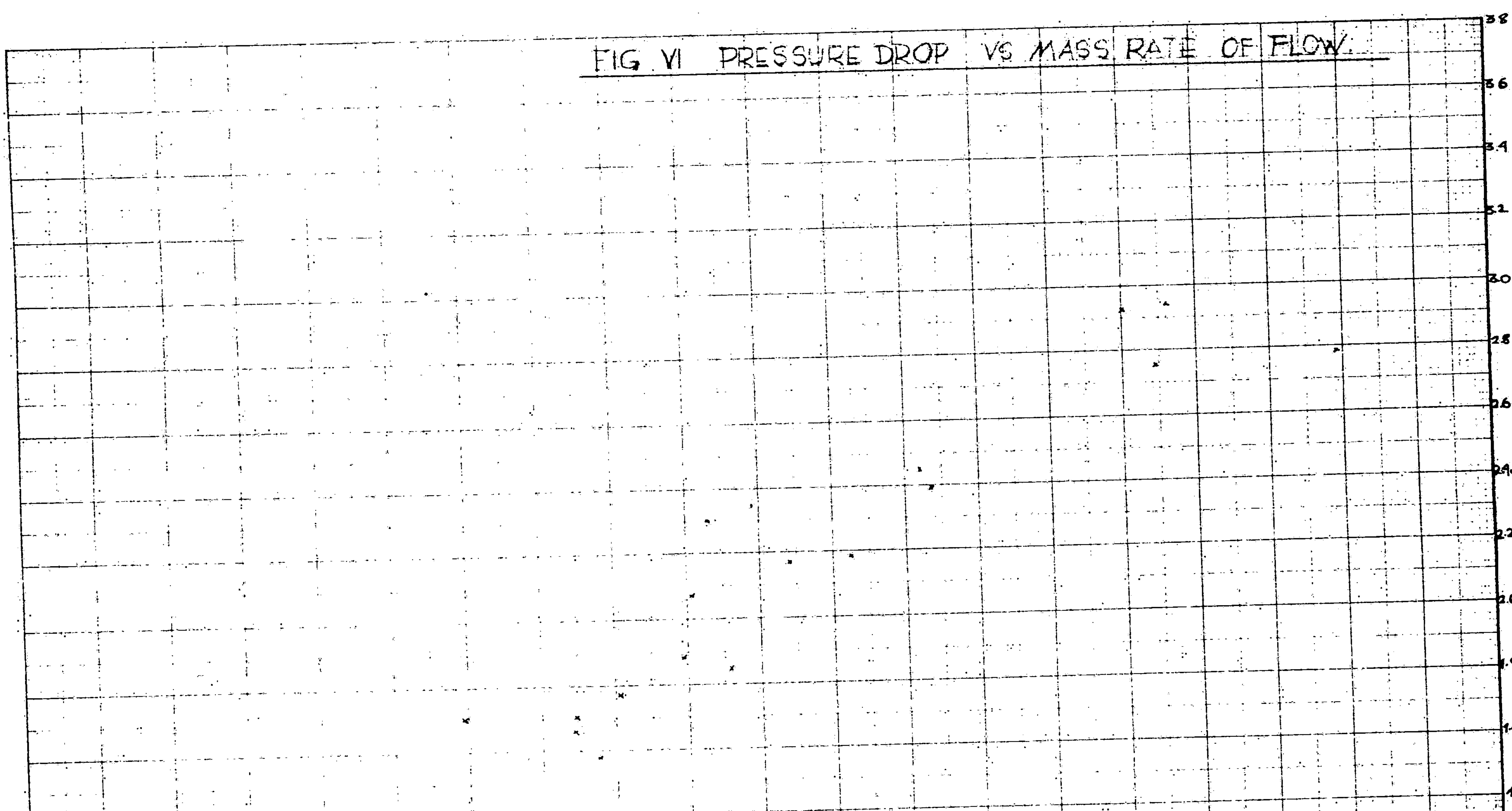


FIG. VI PRESSURE DROP VS MASS RATE OF FLOW.

TABLE III VALUES OF COMPUTED ISENTROPIC AND ACTUAL AMOUNT OF VAPOR FORMED

RUN	PRESSURE TAP	q_{isen}	q_{act}	COMPUTED ISENTROPIC MASS RATE OF FLOW G_{isen}	MEASURED MASS RATE OF FLOW G_{act}
VII	P_1	0.00574	0.000472	457.4	902
	P_5	0.00740	0.000518	483.1	902
	P_7	0.01008	0.000732	497.9	902
	P_9	0.01291	0.001142	548.0	902
13	P_1	0.00555	0.000469	445.8	872
	P_5	0.00684	0.000485	467.5	872
	P_7	0.00892	0.000695	490.5	872
	P_9	0.01266	0.001400	533.3	872
16	P_1	0.00496	0.000381	433.6	872
	P_5	0.00641	0.000438	456.9	872
	P_7	0.00848	0.000820	481.9	872
	P_9	0.01229	0.001080	527.9	872
18	P_1	0.00935	0.000821	509.0	956
	P_5	0.01078	0.000824	525.6	956
	P_7	0.01264	0.000952	546.9	956
	P_9	0.01684	0.001373	584.3	956
19	P_1	0.01113	0.007120	537.7	1110
	P_5	0.01262	0.007130	555.1	1110
	P_7	0.01362	0.007150	564.6	1110
	P_9	0.01755	0.009680	561.5	1110
24	P_1	0.00702	0.000562	478.5	947.5
	P_5	0.00824	0.000563	492.6	947.5
	P_7	0.00968	0.000636	509.1	947.5
	P_9	0.01380	0.001048	553.5	947.5
26	P_1	0.01155	0.000755	614.2	1280
	P_5	0.01321	0.000680	629.1	1280
	P_7	0.01378	0.000684	633.6	1280
	P_9	0.01818	0.000911	666.0	1280
27	P_1	0.000847	0.000525	490.2	1045
	P_7	0.011460	0.000583	526.0	1045
	P_9	0.01608	0.000920	569.7	1045

TABLE III (continued)

RUN	PRESSURE TAP	q_{isen}	q_{act}	COMPUTED ISENTROPIC MASS RATE OF FLOW G_{isen}	MEASURED MASS RATE OF FLOW G_{act}
30	P_1	0.00525	0.000730	373.3	596
	P_5	0.00755	0.001000	395.8	596
	P_7	0.00956	0.001410	421.4	596
	P_9	0.012504	0.002030	449.6	596
31	P_1	0.00363	0.000146	391.5	884
	P_5	0.00585	0.000294	440.6	884
	P_7	0.00820	0.000491	464.0	884
	P_9	0.00987	0.000877	539.0	884

TABLE IV COMPUTED VALUES FOR THE NON DIMENSIONAL GROUPS

RUN	PRESSURE TAP	$\frac{q_{isen}}{q_{act}}$	$(\frac{G_{isen}}{G_{act}})^2$	Re x 10^{-5}	L/D
VII	P_1	12.161	3.880	230,000	43.56
	P_5	14.286	3.486	227,500	61.76
	P_9	11.304	2.719	221,500	71.84
13	P_5	14.103	3.438	216,000	61.76
16	P_1	13.018	4.253	218,000	43.56
	P_5	10.341	3.640	215,500	61.76
	P_9	11.379	2.729	207,000	71.84
18	P_1	11.040	3.527	242,000	43.56
	P_5	13.080	3.318	240,300	61.76
19	P_1	15.630	4.260	280,000	43.56
	P_5	17.700	4.000	277,000	61.76
	P_9	18.440	3.905	269,000	71.84
24	P_1	12.480	3.920	238,000	43.56
	P_5	14.610	3.708	236,500	61.76
	P_9	13.160	2.930	230,000	71.84
26	P_1	15.298	4.343	331,000	43.56
	P_5	19.426	4.137	329,000	61.76
	P_9	19.956	3.690	321,000	71.84
27	P_1	16.133	4,545	262,000	43.56
	P_9	17.478	3.363	250,800	71.84
30	P_1	7.191	2.547	138,000	43.56
	P_5	7.550	2.267	136,200	61.76
	P_9	6.159	1.731	132,000	71.84
31	P_9	11.254	2.686	207,000	71.84

TABLE V COMPUTED VALUES FOR Y vs X

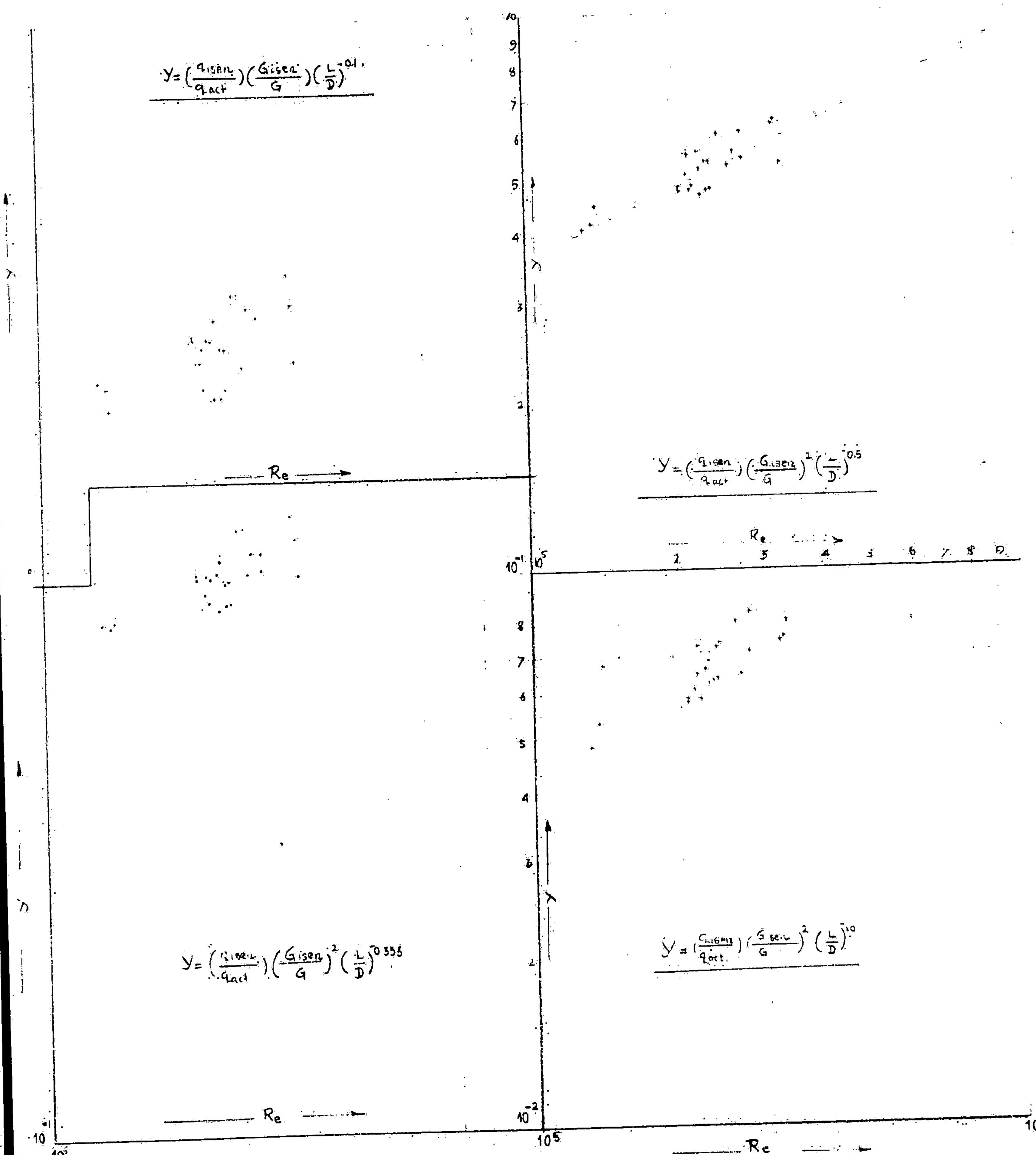
$$\left(\frac{q_{isen}}{q_{act}}\right) = \alpha \left(\frac{L}{D}\right)^{-a} \left(\frac{G_{isen}}{G_{act}}\right)^{2} (R_e)^{-b}$$

or: $$\left(\frac{q_{isen}}{q_{act}}\right) \left(\frac{G_{isen}}{G_{act}}\right)^{2} \left(\frac{L}{D}\right)^{+a} = \alpha \, (R_e)^{-b}$$

where: $$Y = \left(\frac{q_{isen}}{q_{act}}\right) \left(\frac{L}{D}\right)^{+a} \left(\frac{G_{isen}}{G_{act}}\right)^{2}$$

RUN	PRESSURE TAP	FOR $(L/D)^{0.1}$ Y	FOR $(L/D)^{1/3}$ Y	FOR $(L/D)^{1/2}$ Y	FOR $(L/D)^{1.0}$ Y	R_e
30	P_9	2.286	0.8432	0.4135	0.04879	132,000
30	P_5	2.256	0.8424	0.4238	0.05393	136,200
30	P_1	2.051	0.8504	0.4539	0.06868	138,000
16	P_9	2.719	1.0029	0.4919	0.05804	207,000
31	P_9	2.734	1.0083	0.4945	0.05834	207,000
16	P_5	2.470	0.9437	0.5157	0.06041	215,500
13	P_5	2.684	1.0258	0.5605	0.06566	216,000
16	P_1	2.206	0.9148	0.4877	0.07389	218,000
VII	P_9	2.720	1.0026	0.4922	0.05807	221,500
VII	P_5	2.713	1.0379	0.5666	0.06638	227,500
VII	P_1	2.142	0.8871	0.4730	0.06869	230,000
24	P_9	2.928	1.0808	0.5298	0.06251	230,000
24	P_5	2.608	0.9967	0.5447	0.06380	236,500
24	P_1	2.147	0.9051	0.4826	0.07312	238,000
18	P_5	2.617	1.0000	0.5465	0.06403	240,000
18	P_1	2.217	0.9187	0.4899	0.07422	242,000
27	P_9	3.354	1.2372	0.6068	0.07159	250,800
27	P_1	2.433	1.0086	0.5379	0.08148	262,000
19	P_9	3.079	1.1369	0.5571	0.06573	269,000
19	P_5	2.930	1.1196	0.6118	0.07167	277,000
19	P_1	2.515	1.0426	0.5559	0.08422	280,000
26	P_9	3.526	1.3007	0.6379	0.07526	321,000
26	P_5	3.109	1.1878	0.6492	0.07605	329,000
26	P_1	2.414	1.0006	0.5334	0.08082	331,000

FIG: VII PLOT OF Y-VALUES AS A FUNCTION OF R_e FOR SELECTED EXPONENTS FOR $(\frac{L}{D})$



APPENDIX A

Nomenclature:

A = Cross sectional area; ft^2

D = Diameter : ft, if not indicated; O.D. = outside diameter; I.D. = inside diameter.

E = Internal energy BTU/lb.

F = Friction force BTU/lb.

G = Mass flow rate; G_{isen} = mass flow rate based on isentropic expansion : lb/ft^2-sec.

g_c = Conversion factor : 32.4 lb. mass ft/sec^2lb.force.

g = Acceleration due to gravity 32.4 ft/sec^2.

K = Friction factor : dimensionless.

L = Length : ft.

m = Mass : lb.

P = Pressure lb/sq.ft. if not indicated; P_o = pressure in the mixing tank; P_f = pressure in the flash tank; P_1, P_2, etc. subscript referring to the first pressure, second, etc. taps counting from the calming section side.

Q = Heat flow BTU/sec.

q = Vapor formed lb/lb. liquid; q_{isen} = amount of vapor formed if isentropic expansion is assumed; q_{act} = vapor actually formed.

R_e = Reynolds number : dimensionless.

S = Entropy BTU/lb; S_o = entropy of the fluid in the mixing tank; S_x = entropy of the liquid at the point considered; S_{fg} = entropy of evaporation.

T = Temperature ^oF; t = temperature ^oF.

U = Velocity ft/sec.

V = Specific volume; cu.ft/lb; V_f = specific volume of the liquid; V_g = specific volume of the vapor; V_m = average specific volume of the liquid and vapor.

APPENDIX A (continued)

W = External work BTU/lb.

w = Mass rate of flow lb/sec.

Z = Vertical distance from chosen datum plane: ft.

Greek:

ρ = Density lb/cu.ft; ρ_o = density of the fluid in the mixing tank; ρ_x = density of the fluid at the point considered; ρ_a = average density between two points.

λ = Latent heat BTU/lb.

μ = Dynamic viscosity lb/ft-sec.

α = Empirical constant.

γ = Surface tension of the material.

σ = Surface area of the material.

θ = Time : sec.

APPENDIX B

EXAMPLE OF COMPUTING ABSOLUTE PRESSURE FOR DIFFERENT PRESSURE TAPS FROM RECORDED DATA.

RUN 22.

Barometer reading : 29.65
Room temperature : 76°F

Pressure mixing tank :	8.5	8.5	8.5	8.5
Temperature mixing tank :	232°F	232°F	232°F	232° F

Pressure flash tank : +6 . -6

Rate flow : 6.75 Gpm.

Pressure readings : (initial difference : -0.2 +0.2)

P_1	-2.3	+2.15	-2.3	+2.15	-2.3	+2.15	-2.3	+2.15
P_2	-2.45	+2.3	-2.45	+2.3	-2.45	+2.3	-2.45	+2.3
P_3	-2.6	+2.45	-2.6	+2.45	-2.55	+2.4	-2.6	+2.45
P_4	-2.9	+2.75	-2.95	+2.8	-2.9	+2.75	-2.9	+2.75
P_5	-3.3	+3.15	-3.3	+3.5	-3.3	+3.15	-3.3	+3.15
P_6	-6.3	+6.4	-6.25	+6.35	-6.25	+6.35	-6.3	+6.4
P_7	-6.0	+6.1	-5.9	+6.0	-5.9	+6.0	-6.0	+6.1
P_8	-5.6	+5.7	-5.5	+5.6	-5.6	+5.7	-5.6	+5.7
P_9	-4.75	+4.6	-4.75	+4.85	-4.75	+4.85	-4.75	+4.85

Computations:

$$W = 6.75 \text{ Gpm.} = \frac{6.75}{60 \times 7.481} \times \frac{1}{0.01685} \times \frac{4 \times 144}{3.14 \times 0.495^2} = 668.3$$

lb/sec-ft^2

Additional data : Static head = 2.86 mm Hg.

a) $P_f = 29.65 + 2 \times 0.6 = 30.85$ mm Hg.

b) $P_o = 29.65 + (2.036 \times 8.5 - 2.86) = 44.2$ mm Hg.

c) Pressures (absolute) at different pressure taps:

For: P_1 to P_5 : $29.65 + 2.036 \times 8.5 = 0.4 - h$.

Average h from different readings:

P	P	P	P	P	
4.45	4.77	5.02	5.67	6.45	h. av.
42.906	42.586	42.336	41.686	40.906	Abs. Press.

APPENDIX B (continued)

For: P_6 to P_9 : $29.65 - 2.86 + h$

P	P	P	P	P
12.65	12.0	11.25	9.45	h. av.
39.44	38.79	38.04	36.33	Abs. Press.

APPENDIX C

CALCULATION OF ρ BASED ON ISENTROPIC EXPANSION IN THE PIPE AND CONSEQUENT COMPUTATION OF THE MASS FLOW RATE.

Run: 19

Mass flow rate: $1110 \frac{lb.}{sec}$

	P_0	P_1	P_3	P_5	P_7	P_9
S_o	0.3659					
S_x		.3507	.3500	.3486	.3472	.3416
$S_o - S_x$		.0152	.0159	.0173	.0187	.0243
S_{fg}		1.3654	1.3678	1.3702	1.3725	1.3842
$q = \frac{S_o - S_x}{S_{fg}}$		0.01113	0.01162	0.01262	0.01362	0.01755
$1-q$		0.98887	0.98838	0.98738	0.98638	0.90245
V_f	.016992	.016910	.016906	.016899	.016893	.016860
V_g		16.77	16.92	17.2	17.5	18.7
$(1-q)V_f$		0.01672	0.01671	0.01686	0.01666	0.01656
qV_g		0.18886	0.19661	0.21706	0.23842	0.32818
V_m		0.20558	0.21332	0.23392	0.25508	0.34474
ρ_x		4.917	4.687	4.274	3.920	2.900
ρ_o	58.85					
ρ_a		32.883	31.768	31.560	31.385	30.875
$\rho_a \, dP$		158.26	161.50	173.60	190.14	198.40
ρ_o/ρ_a		11.968	12.556	13.769	15.012	20.293
$(\rho_o/\rho_a)^2$		143.233	157.653	189.585	225.36	411.805
$\ln(\rho_o/\rho_a)^2$		4.960	5.048	5.225	5.408	6.000
dP		4.813	5.083	5.501	5.923	6.608
$\frac{\rho_a dP}{\ln(\rho_o/\rho_a)^2}$		31.19	31.99	33.22	34.37	34.00
G_{isen}		537.7	544.7	555.1	564.6	561.5
KL/D		0.06446	0.7941	0.9353	1.0032	1.0629
λ_{act}		0.00712	0.00712	0.00713	0.00715	0.00968

APPENDIX D

TABLE VI -- Values of R_e and K for different mass rate.

Run	G	u x(10+4)		R_e $x10^{-5}$		K
IV	1250	1.582 -	1.635	3.26 -	3.15	0.0143 - 0.145
V	1395	1.607 -	1.632	3.60 -	3.52	0.0139 - 0.0139
VII	902	1.618 -	1.680	2.30 -	2.215	0.0153 - 0.0154
VIII	1061	1.592 -	1.64	2.75 -	2.67	0.0147 - 0.0149
9	809	1.772 -	1.85	1.885 -	1.808	0.0157 - 0.0159
11	589	1.747 -	1.83	1.395 -	1.328	0.0170 - 0.0172
12	652	1.71	1.78	1.571 -	1.51	0.0161 - 0.0166
13	872	1.653	1.724	2.175	2.09	0.0152 - 0.0155
15	634	1.737	1.812	1.508	1.45	0.0165 - 0.0168
16	872	1.655	1.718	2.18	2.095	0.0152 - 0.0155
17	954	1.603	1.657	2.45	2.37	0.0151 - 0.0153
18	965	1.645	1.702	2.42	2.34	0.0152 - 0.0153
19	1110	1.636	1.700	2.80	2.69	0.0147 - 0.0149
20	799	1.631	1.708	2.02	1.93	0.0156 - 0.0159
21	770	1.77	1.85	1.795	1.715	0.0161 - 0.0162
22	657.5	1.712	1.796	1.585	1.51	0.0164 - 0.0166
23	770	1.688	1.77	1.88	1.79	0.0157 - 0.0159
24	947.5	1.642	1.701	2.38	2.30	0.0151 - 0.0152
25	1271	1.61	1.655	3.26	3.17	0.0143 - 0.0144
26	1280	1.592	1.645	3.318	3.21	0.0142 - 0.0143
27	1045	1.647	1.724	2.62	2.508	0.0149 - 0.0150
28	978	1.651	1.720	2.44	2.344	0.0152 - 0.0153
29	1118	1.606	1.652	3.03	2.943	0.0144 - 0.0146
30	596	1.781	1.855	1.38	1.325	0.0170 - 0.0171
31	884	1.67	1.76	2.18	2.07	0.0154 - 0.0155

CALCULATION OF R_e:

Example: Run 20

$$R_e = \frac{D\,G}{u}$$

For: P_1 $u = 1.631 \times 10^{-4}$ $\quad R_e = \frac{0.495 \times 799}{12 \times 1.631 \times 10^{-4}} = 2.02 \times 10^5$

P_5 $u = 1.651 \times 10^{-4}$ $\quad R_e = \frac{0.495 \times 799}{1.651 \times 12 \times 10^{-4}} = 1.995 \times 10^5$

P_7 $u = 1.708 \times 10^{-4}$ $\quad R_e = \frac{0.495 \times 799}{12 \times 1.708 \times 10^{-4}} = 1.930 \times 10^5$

APPENDIX E

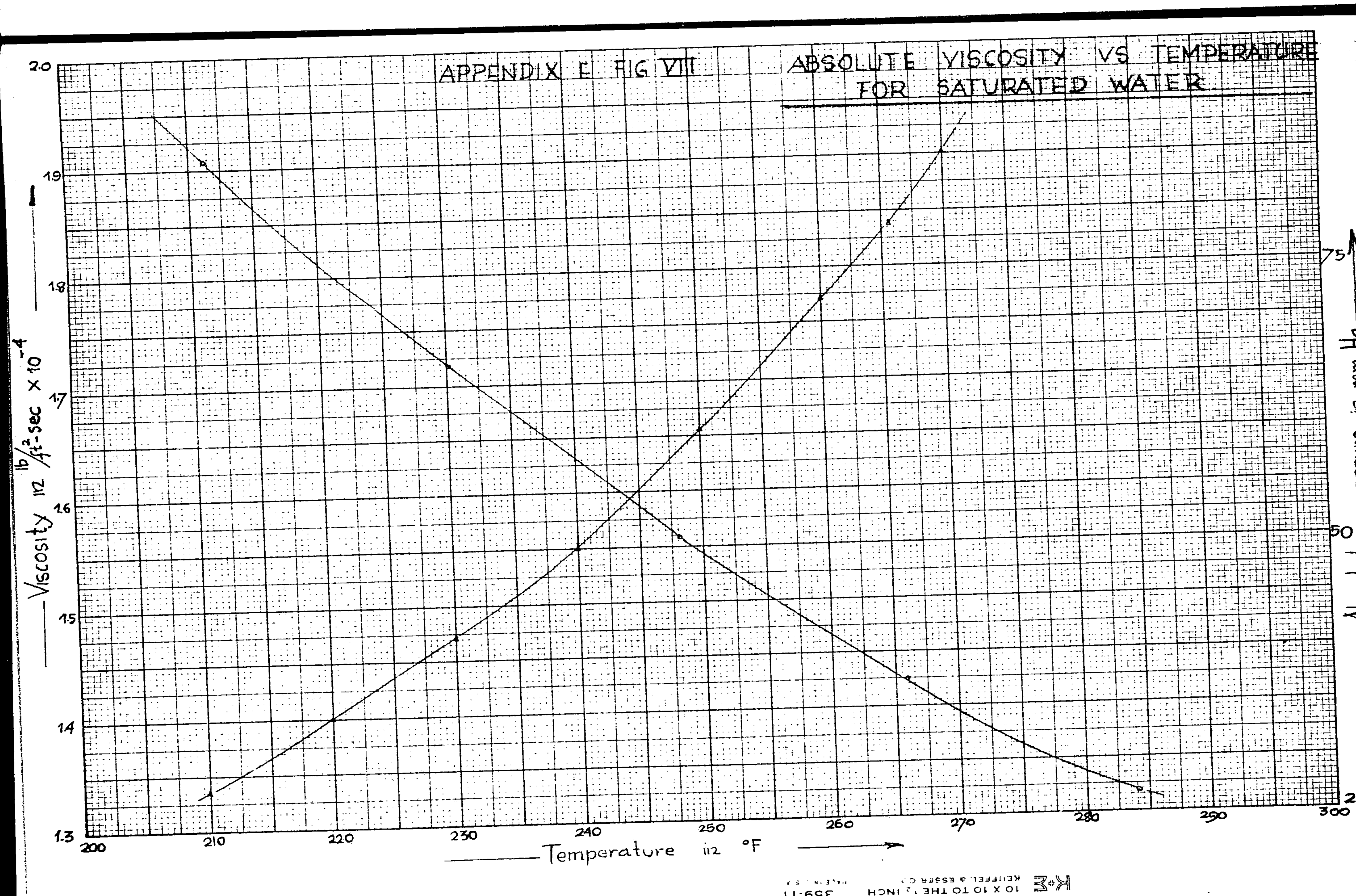
Table VIa - VALUES OF u FOR SATURATED WATER*

Temperature °C	Temperature °F	Pressure (Sat.) Atm	Pressure (Sat.) lb/in^2	Viscosity (Absolute) c.p.$\times 10^{-1}$	Viscosity (Absolute) $lb/ft^2 sec. \times 10^{-4}$
100	212	1.03	14.696	2.83	1.91
125	257	2.37	33.164	2.28	1.535
150	302	4.85	69.046	1.86	1.251
175	347	9.10	129.47	1.58	1.062
200	392	15.9	225.56	1.36	0.915
225	437	26.0	382.1	1.23	0.826
250	482	40.6	596.0	1.13	0.759
275	527	60.7	774.0	1.04	0.695
300	572	87.6	1290.0	0.95	0.638
325	617	123.0	1810	0.84	0.565
360	680	190.4	2708	0.71	0.4775
370	698	214.7	3142	0.63	0.424
374	707	225.2	3308	0.53	0.356

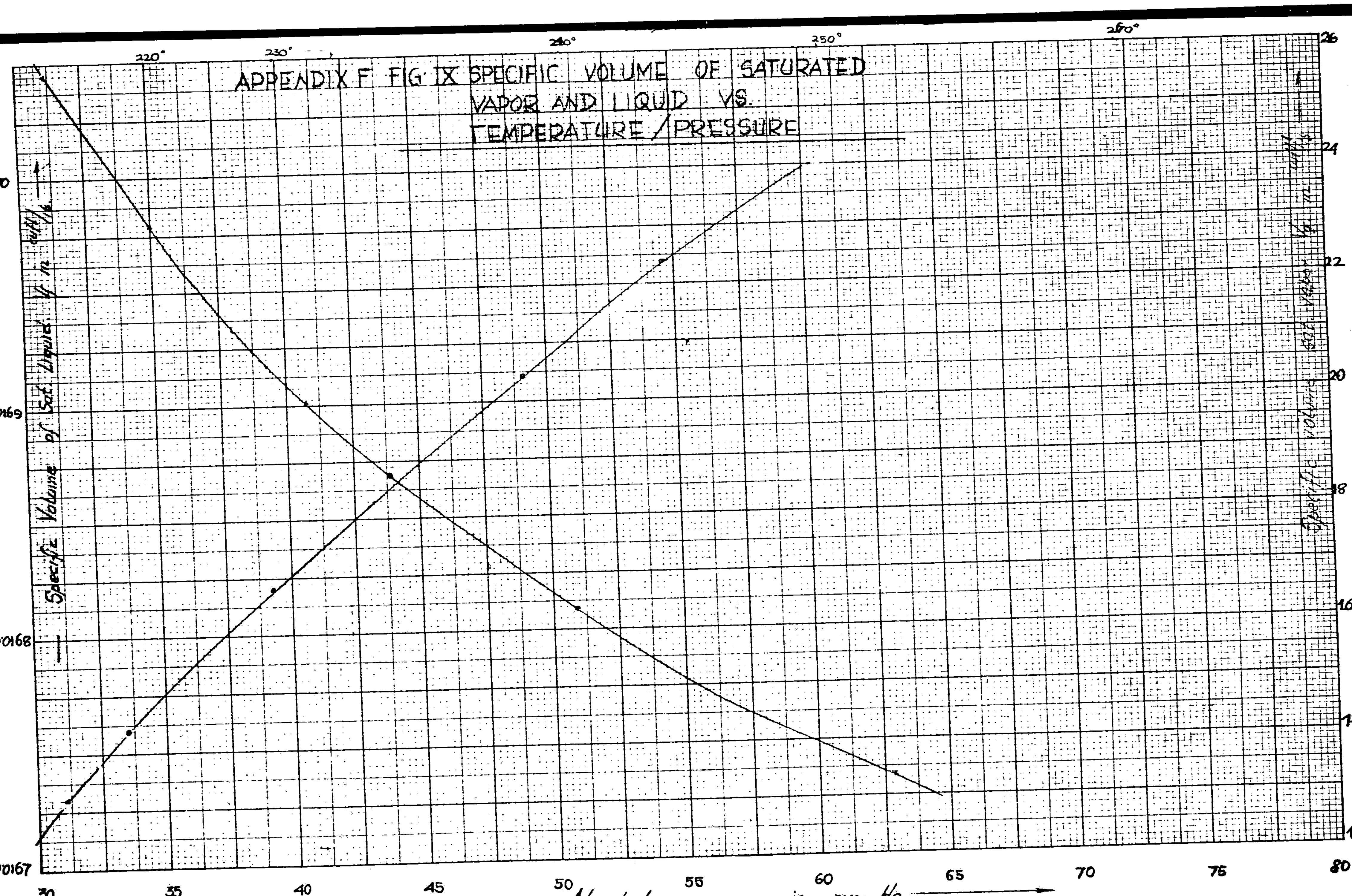
* From K. Sigwart.

Table VIb - VALUES OF u FOR SATURATED WATER**

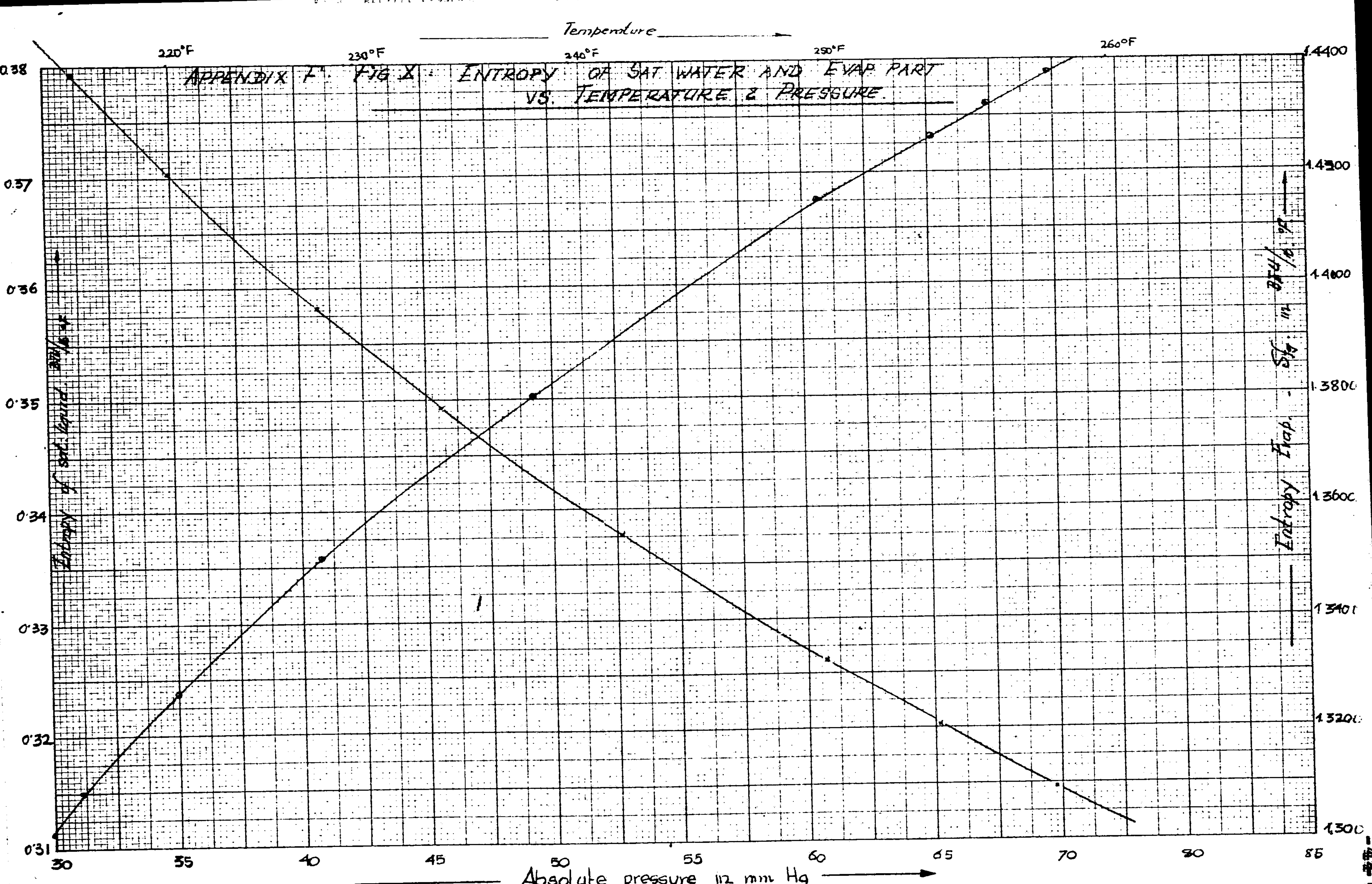
Temperature °C	Temperature °F	Pressure Atm	Pressure lb/in^2	Viscosity (Absolute) c.p.$\times 10^{-1}$	Viscosity (Absolute) $lb/ft^2 sec. \times 10^{-4}$
110	230	1.46	20.78	2.56	1.72
120	248	2.02	28.797	2.32	1.56
130	266	2.75	39.182	2.12	1.425
140	284	3.68	52.418	1.96	1.318
150	302	4.85	69.046	1.84	1.238
160	320	6.30	89.660	1.74	1.17



APPENDIX E FIG VII
ABSOLUTE VISCOSITY VS TEMPERATURE FOR SATURATED WATER
Viscosity in lb/ft-sec × 10^-4
Absolute pressure in mm Hg.
Temperature in °F
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
75
50
25
200
210
220
230
240
250
260
270
280
290
300



APPENDIX F FIG IX SPECIFIC VOLUME OF SATURATED VAPOR AND LIQUID VS. TEMPERATURE / PRESSURE
220°
230°
240°
250°
260°
.00170
.00169
.00168
.00167
Specific Volume of Sat. Liquid V_f in cuft/lb
26
24
22
20
18
16
14
12
30
35
40
45
50
55
60
65
70
75
80
Absolute pressure in mm Hg.



Temperature
220°F
230°F
240°F
250°F
260°F
APPENDIX F. FIG X. ENTROPY OF SAT WATER AND EVAP PART
VS. TEMPERATURE & PRESSURE.
0.38
0.37
0.36
0.35
0.34
0.33
0.32
0.31
Entropy of sat. liquid BTU/lb °F
1.4400
1.4200
1.4000
1.3800
1.3600
1.3400
1.3200
1.3000
Entropy Evap - S_{fg} in BTU/lb °F
30
35
40
45
50
55
60
65
70
80
85
Absolute pressure in mm Hg

APPENDIX G

CALCULATION OF VALUES OF EXPONENT FOR R_e

$$y = \alpha x^a$$

$$y = \left(\frac{q_{isen}}{q_{act}}\right) \left(\frac{L}{D}\right)^{0.5} \left(\frac{G_{isen}}{G}\right)^2$$

$$x = R_e$$

$$\Sigma \log y_1 = \Sigma \log \alpha + a \Sigma \log x_1$$

$$\Sigma \log y_2 = \Sigma \log \alpha + a \Sigma \log x_2$$

$$(\Sigma \log y_1 - \Sigma \log y_2) = a\,(\Sigma \log x_1 - \Sigma \log x_2)$$

$$a = \frac{\Sigma \log y_1 - \Sigma \log y_2}{\Sigma \log x_1 - \Sigma \log x_2}$$

y	log y	x	log x
0.4135	-0.383471	132,500	5.122216
0.4238	-0.372878	136,200	5.134177
0.4534	-0.343568	138,000	5.139879
0.4919	-0.308110	207,000	5.315970
0.4945	-0.305804	207,000	5.315970
0.5157	-0.287586	215,500	5.333447
0.5005	-0.251389	216,000	5.334454
0.4877	-0.311864	218,000	5.338456
0.4922	-0.307853	221,500	5.345374
0.5666	-0.246711	227,500	5.356981
0.4730	-0.325154	230,000	5.361728
0.5298	-0.275888	230,000	5.361728
$\Sigma \log y_1 =$ -3.720276			$\Sigma \log x_1 =$ 63.460380

y	log y	x	log x
0.5447	-0.263886	236,500	5.372912
0.4826	-0.316463	238,000	5.376577
0.5465	-0.262389	240,000	5.380754
0.4899	-0.309912	242,000	5.383815
0.6068	-0.216950	250,800	5.411620
0.5379	-0.269326	262,000	5.418301
0.5571	-0.254048	269,000	5.429752
0.5559	-0.255044	280,000	5.447158
0.6379	-0.195227	321,000	5.506505
0.6118	-0.213370	277,000	5.442480
0.5334	-0.272933	331,000	5.519828
0.6492	-0.187638	329,000	5.517196
$\Sigma \log y_2 =$ -2.817186			$\Sigma \log x_2 =$ 65.706416

APPENDIX G (continued)

$$a = \frac{-3.720276 - (-2.817186)}{63.460380 - 65.706416} = \frac{-0.903090}{-2.247536} = 0.40$$

The coefficient α can be found as follows:

$$-3.720276 = 12 \log \alpha + 0.40 \times 63.460380$$

$$\log \alpha = \frac{-29.104424}{12} = -2.425368$$

$$= 0.574632 - 3$$

$$\alpha = 0.00375$$

REFERENCE

1. Benjamin, M. W. & Miller, J. G., Transactions A.S.M.E. vol. 63 pg. 419-426, 1941.

2. Benjamin, M. W. & Miller, J. G., Transactions A.S.M.E. vol. 64 page 657-669, 1942.

3. Bottomley, W. T., Trans. N.E.C. Inst., vol. 53 page 65 (1936-1937).

4. Burnell, R., Journ. Inst. Engrg. of Australia, vol. 18 reprinted in Engineering, vol. 164 page 572-576, 1947.

5. Heidweiler, Ann. der Physik, vol. 59 page 193-212, 1896.

6. Isbin, Discussion -- see reference 10.

7. Keenan & Keyes, Thermodynamic Properties of Steam.

8. Ledinegg, Mächinenbau und Warmewirtschaft, vol. 3, No. 4 page 45-47, 1948.

9. Profos, P., Schweizer Archiv., vol. 13, No. 1 page 1-8, 1947, reprinted in Mech. World 1947.

10. Schweppe, J. L., & Foust, A. S., Reprints of Heat Transfer Symposium, page 242-291, A.I.Ch.E. 1951.

11. Sigwart, K., Forschung Gebiete der Ingenieurswesens, vol. 7 page 125-140, 1936.

12. Silver, R. S., Proceedings Inst. of Mech. Engrg., vol. 153 page 261-281, 1945.

13. Silver, R. S., Proceedings of the Royal Society, vol. 194 A, page 464-480, 1948.

14. Rateau, Ann. des Mines (1902).

AUTHOR'S VITA

Name: Sugeng Sundjaswadi

Place of birth: Surabaia, Indonesia

Date of birth: 26th of March 1921

Education:
a. Elementary School, Surabaia, 1928-1935.
b. 1st Public High School, Surabaia, 1935-1940.
c. College of Engineering and Technology, Jadavpur, Calcutta, India, 1947-1951.
d. University of Sheffield, Sheffield, Great Britain, 1952-1953.
e. Lehigh University 1953-1955.

Degree obtained: Bachelor of Chemical Engineering, College of Engineering and Technology, Jadavpur, India, 1951.

Other certificates: Certificate, stating that the author has passed interdepartmental examination in Fuel Engineering, University of Sheffield, Department of Fuel Engineering, 1953.